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EFFECT OF GLOBAL SOLAR RADIATION ON THE HEAT TRANSFER AND PRESSURE DROP CHARACTERISTICS OF DOUBLE FLOW PACKED BED SOLAR AIR HEATER

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ABSTRACT

An analytical study was conducted to evaluate the thermal performance of a new solar air heater using wire mesh packed bed in double flow. Using the first law of thermodynamics the energy balance equation were written and the expression for temperature rise and thermal efficiency have been developed. Results showed that the global solar radiation shows an enhancement in the temperature rise through the packed bed double flow solar air heater and this is a strong function of mass flow rates. An enhancement of 3.43 to 4.34 times has been found in temperature rise and 4 to 5% enhancement in thermal efficiency for the investigator range of global solar radiation and mass flow rate. Also the pressure drop has been found independent of global solar radiation.

KEYWORDS: Solar Air Heater, Packed Bed, Thermal Efficiency, Temperature Rise & Global Solar Radiation

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Original Article

1. INTRODUCTION

Solar air heating is a solar thermal technology in which the energy from the sun is captured by an absorbing medium and used to heat air. Solar air heater is a renewable energy heating technology used to heat or condition air for the building or process heat application. It is typically the most cost–effective out of all the solar technologies especially in commercial and industrial applications, and it addresses the largest usageof building energy in heating climates, which is space heating and industrial process heating. Sharma et al. [1],described the design and performance of a matrix type solar air heater. They presented a transient model for a matrix air heater having a single glazing at the top and a thin metallic plate backed by an insulator at the bottom. Analytical investigation with the experimental validation were presented for the variation of outlet air temperature. Dhiman et al. [2], developed an analytical model of a novel parallel flow packed bed solar air heater with packed material in its upper channel to predict its thermal performance. They studied the effects of mass flow rate of air and porosities of the packing material on the thermal performance and reported that a parallel flow packed bed solar air heater exhibits better performance than the conventional non-porous double flow solar air heater with 10-20% increase in its thermal efficiency. Dhiman and Singh [3], established an analytical model to predict the thermal performance of two different designs of double pass packed bed solar air heater under external recycle. Mittal and Varshney [4], investigated a packed bed solar air heater having its duct packed with blackened wire screen matrices of different geometrical parameters. They determinedthermohydraulic performance in term of effective efficiency on the basis of actual thermal energy gain subtracted by the primary energy required to generate power needed for pumping air through the packed bed. Ozgen et al.[5], investigated experimentally on the thermal performance of a double flow

solar air heater having aluminium cans and reported that by inserting aluminium cans on absorbing plate into the double pass channel in a flat plate solar air heater, gets improved the collector efficiency due to enhancements in heat transfer coefficient between the absorber plate and air. Prasad et. al.[6], made an experimental investigation on a packed bed solar air heater using wire mesh as packing material and compared with conventional solar air heater. They reported that the thermal efficiency of a packed bed solar air heater using wire mesh as packing material can be increased up to 76.9 to 89.5% more than the conventional one. Ramadan et. al.[7]studied analytically and experimentally the thermal performance of a double–pass solar air heater with a packed bed above the heater absorber plate and found that the thermohydraulic efficiency increased with the increase in mass flow rate until a typical value of 0.05 kg/s, beyond which the increase in thermohydraulic efficiency becomes insignificant. They concluded that to operate such a system the value of mass flow rate equal to 0.05 kg/s or lower leads to lower pressure drop across the system would be more advantageous. Sharma et al.[8] investigated experimentally the thermal performance of a single flow solar air heater having its duct packed with blackened wire screen matrices. They measured that thermal performance of plane collector and found appreciable improvement in the performance by packing its duct with blackened wire-screen matrices and this improvement was a strong function of bed and operating parameters. Singh et al.[9] studied the thermal performance of packed bed heat storage system for solar air heaters. Wijeysundera et al.[10] studied the operation of a conventional solar air heater with two covers in a two pass mode. They observed an enhancement of the collector efficiency by 10-15% for various operating conditions. Yeh et al.[11] investigated experimentally the collector efficiency of double flow solar air heaters with fins attached over and under the absorber plate. Hernandez and Quinonez [12] developed two analytical model which describe the thermal behaviour of solar air heaters of double parallel flow and double pass counter flow. Choudhury and Garg.[13] conducted comparative theoretical parametric analysis of solar air-heating collectors with and without packing in the flow passage above the back plate. They concluded that smaller diameter and lower porosity of pickings, shallower duct depth, longer air channels, and larger air mass flow rates result in higher efficiency with higher pressure drop and hence larger fan running costs of the systems. Prasad and Saini [14] investigated experimentally the thermal performance of wire-screen matrix bed solar air heaters of unidirectional flow and cross flow types and a plane collector. They reported that under similar conditions both the unidirectional flow and cross flow matrix bed collector have optimum thermal efficiency at an optical depth of bed nearly 5. Singh and Sharma [15] made theoretical parametric analysis of double flow solar air heater having expanded metal mesh as artificial roughness on both side of the absorber plate. Theyapplied the effect of the friction in the upper and lower flow channels and geometrical parameters of the roughness on thermo- hydraulic efficiency of the collector. Ong[16] presented a theoretical model for predicting the thermal performance of four different types of single pass solar collector. They found that decreasing the wind heat transfer coefficient,the overall heat loss coefficients were appreciably increased. Sodha et.al.[17] in his numerical work presented the double flow non- porous solar air heater. They investigated the effect of mass flow rate and collector length and found that double flow air heater performed more efficiently than a single flow air heater. Chiiet al. [18] investigated experimentally and theoretically on the performance of wire mesh packed double pass air heater found an under external recycle. They found the effect of recycle ratio on the heat transfer efficiency and enhancement as well as power consumption increments has been also declined. Yeh et al. [19] investigated theoretically and experimentally the effect of the fraction of mass flow rate on the performance of double flow solar air heater. They found that for same mass flow rate, double flow solar collector performs more efficiently than single pass solar air heater. Paisarn [20] studied the heat transfer characteristics and performance of double pass flat plate solar air heater with and without porous media. They found the solar collector with porous media gives 25% higher thermal

efficiency than without porous media. Sopian etal.[21] studied the effect of mass flow rate, solar radiation and thermal efficiency of double pass solar collector. The study concluded that the presence of porous media in the channel increases the outlet temperature and the thermal efficiency of the system.

The above literature revealed that that the double flow packed bed solar air heater has been investigated properly although performance enhancement can take place. So, in the present paper, double flow packed bed solar air heater has been investigated analytically. The effect of global solar radiation on the thermal performance has been done by developing the energy balance equation and solving then to determine the thermal performance.

2. THEORETICAL ANALYSIS

A double flow solar air heater its upper duct packed with wire screen, as shown in Figure.1has been considered. The channel of the width (W), depth (H) and length (L) and having one glass cover, is uniformly heated from top by intensity of radiation transmitted through the glass cover.

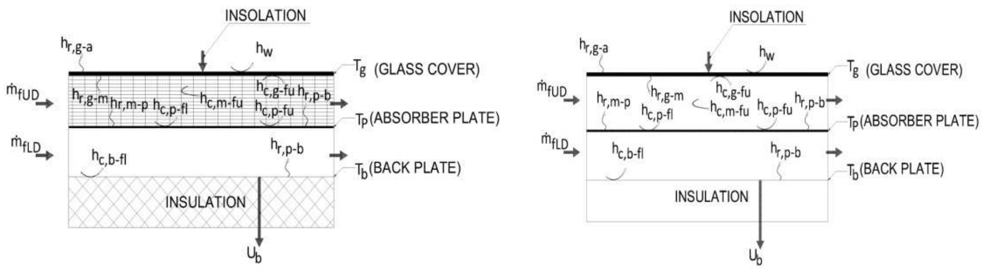

Figure 1: Schematic Diagram of (A) DFPBSAH and (B) The Energy Balance of DFPBSAH.

These radiations heat the matrix material, which in turn heats the air flowing through the duct at higher temperature. In this study the combined effect of wire screen packed bed in double flow has been considered. The energy balance equations for different elements have been written and solved using numerical technique.

2.1: Energy Balance Equations

The energy balance equations for different components of solar air heater under consideration are presented below based on the following assumptions:

- The system operates under steady state condition.
- The heat capacities of the glass cover, absorber plate, back plate and insulation are negligible.
- The flow is one dimensional i.e. the temperature of the flowing air varies only in the direction of flow (x-direction).
- Thermo-physical properties of the flowing air are assumed to be varying linearly with temperature.
- There is no temperature gradient across the thickness of the glass covers, absorber and back plates.

Glass Cover

$$I\alpha_g = \left(h_{r,g-a}+h_w\right)\left(T_g-T_a\right)+h_{r,g-m}\left(T_g-T_m\right)+h_{c,g-fu}\left(T_g-T_{fu}\right) \quad (1)$$

Packed Bed

$$I\tau_g \alpha_m A_m = h_{r,m-g}(T_m - T_g)A_g + h_{r,m-p}(T_m - T_p)A_P + h_{c,m-fu}(T_m - T_{fu})A_m \quad (2)$$

Air Flowing in Upper Duct

$$h_{c,m-fu}(T_m - T_{fu})A_m + h_{c,p-fu}(T_P - T_{fu})A_p + h_{c,g-fu}(T_g - T_{fu})A_g + 2\dot{m}_{fu}C_p(T_{fu} - T_i) \quad (3)$$

Absorber Plate

$$h_{r,m-p}(T_m - T_p)A_m = h_{r,p-b}(T_p - T_b)A_P + h_{c,p-fu}(T_p - T_{fu})A_P + h_{c,p-fl}(T_p - T_{fl})A_p \quad (4)$$

Air Flowing in Lower Duct

$$h_{c,p-fl}(T_p - T_{fl})A_p + h_{c,b-fl}(T_b - T_{fl})A_b = 2\dot{m}C_p(T_{fl} - T_{fi}) \quad (5)$$

Bottom Plate

$$h_{r,p-b}(T_p - T_b)A_p = h_{c,b-fl}(T_b - T_{fl})A_b + U_b(T_b - T_{fl})A_b \quad (6)$$

Rearranging Eqs. (1-6) gives the mean temperatures of glass, packed bed, air in upper duct, absorber plate, air in lower duct and bottom plate respectively as follow:

$$T_g = \frac{I\alpha_g + \left(h_{r,g-a} + h_w\right)T_a + \left(h_{r,g-m}\right)T_m + \left(h_{c,g-fu}\right)T_{fu}}{\left(h_{r,g-a} + h_w\right) + \left(h_{r,g-m} + h_{c,g-fu}\right)} \quad (7)$$

$$T_m = \frac{A_m I\tau_g \alpha_m + h_{r,m-g}T_g A_g + h_{r,m-p}T_p A_P + h_{c,m-fu}T_{fu}A_m}{h_{r,m-g}A_g + h_{r,m-p}A_p + h_{c,m-fu}A_m} \quad (8)$$

$$T_{fu} = \frac{h_{c,m-fu}T_m A_m + h_{c,p-fu}A_p T_P + h_{c,m-fu}A_g T_g + 2\dot{m}_{fu}c_p T_{fu}}{h_{c,m-fu}A_m + h_{c,p-fu}A_P + h_{c,m-fu}A_g + 2\dot{m}_{fu}c_p} \quad (9)$$

$$T_P = \frac{h_{r,m-p}T_m A_m + h_{r,p-b}T_b A_P + h_{c,p-fu}A_P T_{fu} + h_{c,p-fl}A_P T_{fl}}{h_{r,m-p}A_m + h_{r,p-b}A_P + h_{c,p-fu}A_p + h_{c,p-fl}A_p} \quad (10)$$

$$T_{fl} = \frac{h_{c,p-fl}A_p T_P + h_{c,b-fl}A_b T_b + 2\dot{m}c_p T_i}{h_{c,p-fl}A_p + h_{c,b-fl}A_b + 2\dot{m}c_p} \quad (11)$$

$$T_b = \frac{h_{r,p-b}T_p A_P + h_{c,b-fl}A_b T_{fl} + U_b A_b T_a}{h_{r,p-b}A_p + h_{c,b-fl}A_b + U_b A_b} \quad (12)$$

2.2 Evaluation of Heat Transfer Coefficients

The various heat transfer coefficients given in above equations can be calculated from the following empirical relations [20] and [22] as,

$$h_{r,g-a} = \frac{\sigma\left(T_g^2 + T_a^2\right)\left(T_g + T_a\right)}{\frac{1}{\varepsilon_g} - 1} \quad (13)$$

$$h_{r,g-m} = \frac{\sigma\left(T_g^2 + T_m^2\right)\left(T_g + T_m\right)}{\left(\frac{1}{\varepsilon_g} + \frac{1}{\varepsilon_m} - 1\right)} \quad (14)$$

$$h_{r,m-p} = \frac{\sigma\left(T_m^2 + T_p^2\right)\left(T_m + T_P\right)}{\left(\frac{1}{\varepsilon_m} + \frac{1}{\varepsilon_P} - 1\right)} \quad (15)$$

$$h_{r,p-b} = \frac{\sigma\left(T_P^2 + T_b^2\right)\left(T_P + T_b\right)}{\left(\frac{1}{\varepsilon_P} + \frac{1}{\varepsilon_b} - 1\right)} \quad (16)$$

Convective heat transfer coefficient due to wind velocity can be given by the relation [23]

$$h_w = 5.7 + 3.8Vw \quad (17)$$

For wire mesh screen matrix bed, packing to air heat transfer coefficient $h_{c,m-fu}$ is calculated by correlations [4], given below

$$h_{c,m-fu} = j_H\left(\mathrm{Pr}\right)^{-2/3} G_o C_p \quad (18)$$

where $j_H = 0.647\left[\frac{1}{n\varphi}\left(P_t/d_w\right)\right]^{2.104} \mathrm{Re}_m^{-0.55}$ (19)

Convective heat transfer coefficient between the air flowing in the packed duct and the glass cover, $h_{c,g-fu}$ may be obtained as [2],

$$h_{c,g-fu} = \frac{Nu_m k_{fu}}{4r_h} \quad (20)$$

The Nusselt number Nu_m for the wire screen packed bed is given by [25]

$$Nu_m = 0.2\,\mathrm{Re}_m^{0.8}\,\mathrm{Pr}^{1/3} \quad (21)$$

where Re_m is the Reynold's number for the wire mesh packed bed duct and is calculated as [25]

$$\mathrm{Re}_m = 4r_h \cdot \frac{G_o}{\mu} \tag{22}$$

$$r_h = \frac{\phi d_w}{4(1-\phi)} \tag{23}$$

Where r_h is hydraulic radius of the upper duct and is related to the packing size and the void space. G_o is the mass velocity and, is given by

$$G_o = \frac{\dot{m}_{fu}}{A_c \phi} \tag{24}$$

ϕ is the porosity of the wire screen packed bed and is given by [25]

$$\phi = \frac{P_t^2 H - \left[\frac{\pi}{2}(d_w)^2 P_t\right] n}{P_t^2 H} \tag{25}$$

where **n** is the number of wire screen layers and H is the height of the duct.

The convective heat transfer coefficient between the absorber plates to the air flowing in the upper duct $h_{c,p-fu}$ is assumed to be equal to $h_{c,g-fu}$ [2]

Convective heat transfer coefficient for flow in lower duct can be calculated as follows:

$$h_{c,p-fl} = h_{c,b-fl} = \frac{Nu_{pb} K_{fl}}{D_{hl}} \tag{26}$$

where, D_{hl} is the hydraulic diameter for the lower duct and is given as

$$D_{hl} = \frac{2(WH)}{(W+H)} \tag{27}$$

The value of Nusselt Number Nu_{pb} can be calculated from the relationship, according to the type of flow in the duct, [24] as given below

(a) For Laminar Flow (Re<2300)

$$Nu_{pb} = 5.4 + \frac{0.00190\left[\mathrm{Re}\,\mathrm{Pr}\left(\frac{D_{hl}}{L}\right)^{1.71}\right]}{1 + 0.00563\left[\mathrm{Re}\,\mathrm{Pr}\left(\frac{D_{hl}}{L}\right)^{1.71}\right]} \tag{28}$$

(b) For Transition Flow (2300<Re<6000)

$$Nu_{pb} = 0.116\left(\mathrm{Re}^{2/3} - 125\right)\mathrm{Pr}^{2/3} + \left[1 + \left(\frac{D_{hl}}{L}\right)^{\frac{2}{3}}\right]\left(\frac{\mu}{\mu_w}\right)^{0.14} \tag{29}$$

(c) For Turbulent Flow (Re>6000)

$$Nu_{pb} = 0.018\,\mathrm{Re}^{0.8}\,\mathrm{Pr}^{0.4} \tag{30}$$

2.3: Performance Parameters

The useful heat gain, Qu, of the collector is given as

$$Qu = Qu_{fu} + Qu_{fl} \tag{31}$$

Where,

$$Qu_{fu} = 2\dot{m}_{fu} C_P (T_{fu} - T_i) \tag{32}$$

$$Qu_{fl} = 2\dot{m} f_l C_P (T_{fl} - T_i) \tag{33}$$

Thermal efficiency of the solar air heater is calculated as:

$$\eta_{th} = \frac{Q_u}{IA_c} \tag{34}$$

The detailed procedure for the calculation of thermal efficiency is shown in figure.2 in the form of flow chart.

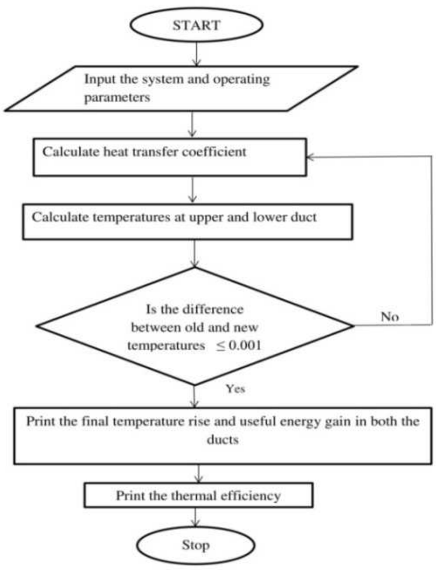

Figure 2: Flow Chart for Calculation of Thermal Performance of Double Flow Packed Bed Solar Air Heater

3. RESULTS AND DISCUSSIONS

The performance parameters of the double flow packed bed solar air heater have been computed for the range of relevant parameters given in table 1. Among these parameters, some have constant values while some vary in the specific range as per requirements. Usually the air temperature at the outlet of the solar air heater duct has been found to be in the range of 35 - 40°C above the ambient, and for ambient.

Table.1 Range of Values of Parameters Used in Analytical Computation

Parameters	Value (Range)	Parameters	Value (Range)	Parameters	Value (Range)
L	1.2 m	r_c	0.076	K_f	0.02673 W/m-k
b	1m	t_r	0.838	μ	1.8658×10^{-5} kg/s-m
a_P	1.2 cm	K_s	62.73 W/m-k	m	0.0164-0.0362 kg/s
β	$600\,m^{-1}$	C_p	1005 J/kg-k	U_t	7-14.5 W/m^2-k
P_t	2.54 cm	T_a	27^0 C	U_b	0.8 w/m^2-k
T_{fi}	30^0C	$\varepsilon_g=\varepsilon_P$	0.80	I	350-1100 w/m^2

To study the effect of global solar radiation on thermal, performance it has been varied from 350 to 1100 W/m^2. Figure.3 shows the effect of global solar radiation on rise in air temperature for various mass flow rates. It is seen from plot that the air temperature rise increases linearly as the heat flux increases for all mass flow rates. A maximum enhancement of 3.43 times and 4.34 times in temperature rise have been achieved by varying global solar radiation from 350 to 1100 W/m^2 at mass flow rates 0.0164 and 0.0362 kg/s respectively. It may because the increase in global solar radiation increase the bed temperature that corresponds to the maximum heat transfer and rise in air temperature.

The effect of global solar radiation on thermal efficiency for various values of mass flow rate is shown in Figure.4. It clearly indicates that the variation of global solar radiation has considerable effect on thermal efficiency at all mass flow rates. This is because global solar radiation has more effect on convective heat transfer between the bed, absorber plate and air and a little effect on convective heat transfer between absorber plate and glass cover. Also, increase in global solar radiation increase the temperature rise but the efficiency depends on the ratio of rise in temperature to the global solar radiation, thus the dependency on numerator and denominator leads to enhancement in thermal efficiency as the ratio of rise in temperature and global solar radiation increases. In the double flow solar air heater, thermal efficiency increases from 52.4% to 59.62% at lower global solar radiation the thermal efficiency increases from 55.2% to 61.94%.

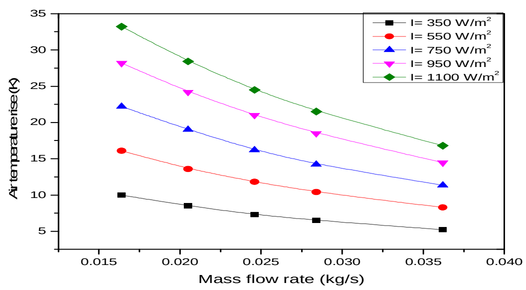

Figure 3: Variation of Air Temperature Rise for Various Values of Global Solar Radiation.

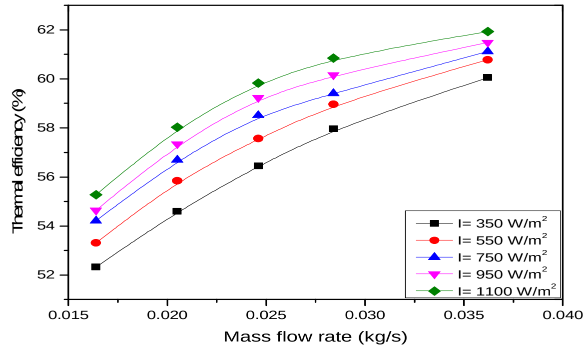

Figure 4: Variation of Thermal Efficiency for Various Values of Global Solar Radiation

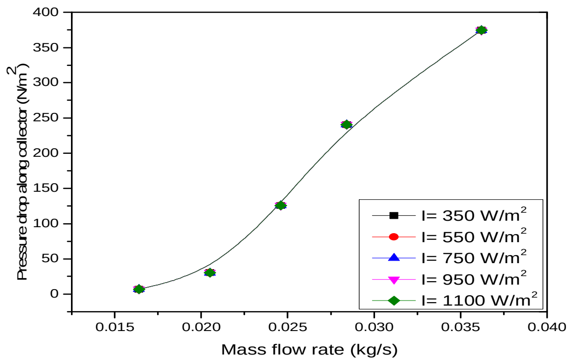

Figure 5: Variation of Pressure Drop Along Collector for Various Values of Global Solar Radiation.

Figure. 5 shows the variation of pressure drop along collector for various mass flow rates and heat flux. It is clearly shown that the pressure drop is independent of global solar radiation.

4. CONCLUSIONS

Based on the above results and discussion following conclusion may be drawn:

- The energy balance equation have been developed to find the expression for thermal efficiency and temperature rise.
- The effect of heat flux on air temperature rise shows a maximum enhancement of 3.43 times and 4.34 times for the heat flux variation from 350 to 1100 W/m^2 with mass flow rate 0.0162 to 0.0362 kg/s.
- For the lower value of global solar radiation of 350 W/m^2 the thermal efficiency increased from 52.4 % to 59.62% and for the higher value of global solar radiation, the thermal efficiency increased from 55.2% to 61.94%.
- For the all values of global solar radiation, pressure drop is independent.

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